COLD-AIR INVESTIGATION OF A TURBINE WITH STATOR-BLADE TRAILING-EDGE COOLANT EJECTION

III - OVERALL STAGE PERFORMANCE

by Edward M. Szanca, Harold J. Schum, and Herman W. Prust, Jr.

Lewis Research Center

SUMMARY

A cold-air experimental investigation was conducted on a 30-inch (0.762-m) single-stage turbine to determine the effect on performance of coolant flow ejection from the stator blade trailing edges. The hollow-cored stator blades of a previously tested turbine were modified to provide the coolant-ejection slots along the trailing edges. The effect of this modification was determined by testing the turbine with zero coolant flow and comparing the results with those previously reported for the same turbine before the modification. The effect of coolant was determined by testing over a range of coolant fraction (ratio of coolant-to-primary flow) from zero to 0.07. The effects of coolant addition were also compared with those predicted analytically.

It was found that the stator-blade modification had little effect on turbine performance. At equivalent design speed and at an equivalent specific turbine work output of 17.00 Btu per pound (39 572 J/kg), the efficiency of the modified uncooled turbine was 0.920. The unmodified turbine efficiency was 0.923. At the same turbine operating point and with the coolant inlet pressure equal to the turbine inlet pressure (30 in. of mercury absolute (1.0159×10^5 N/m^2)), the turbine yielded an efficiency (based on primary air) of 0.958. The attendant coolant fraction was 0.0468.

At a specific turbine operating point, the experimental values of turbine primary-air efficiency increased with coolant fraction. This change reflected the manner of increasing coolant flow by increasing its inlet pressure (and specific energy) relative to that of the primary flow. When the turbine was charged with the ideal energy of the coolant flow, the thermodynamic efficiency varied less than 1 percent (±0.007) over the coolant-fraction range investigated. At low coolant fractions the efficiency first increased due to a recovery of trailing-edge losses as effected by coolant ejection into the wake. As coolant fraction was increased, the frictional losses within the blade became predominant, resulting in the thermodynamic efficiency decrease. Experimentally obtained changes in turbine work output with coolant fraction agreed well with that predicted theoretically.

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INTRODUCTION

Gas turbine engines used in advanced types of aircraft must use high turbine-inlet temperatures in order to meet their mission objectives. These temperatures, in general, are sufficiently high as to require cooling of the turbine blading. The cooling method most commonly considered uses air that is bled from the compressor, ducted through the cooling passages of the blades, and then discharged into the turbine gas stream. The turbine blading for this type of application is characterized by thick profiles and blunt leading and trailing edges. These geometric features result from the necessity of incorporating the blade cooling passages within the profile.

A part of the current turbine research program at NASA Lewis is concerned with the effect on turbine aerodynamic performance of discharging coolant flow into the main gas stream. This effect is related to the manner in which the coolant is ejected. In some cooled blade designs, coolant is ejected from the blade trailing edges. Accordingly, the effect on turbine performance of ejection of coolant flow through stator-blade trailing-edge slots is being investigated. An existing stator was provided with slots, and two separate investigations of this stator have been made without the rotor in place. Results of the first investigation, which was concerned with the effect of coolant flow on overall stator performance, is reported in reference 1. Reference 2 presented the results of detailed radial and circumferential total-pressure measurements at the exit of the slotted stator, both with and without coolant flow. These studies indicated that (1) with no coolant flow, no significant change in stator efficiency resulted from modifying the original blades to incorporate the trailing-edge slots, and (2) the stator efficiency (based on primary air-mass flow) increased with coolant flow.

The purpose of the investigation reported herein was to determine the net effect on turbine performance of both the stator modification and coolant flow. Performance characteristics of the modified turbine with no coolant flow were obtained over a range of speed and pressure ratio and compared with those obtained for the unmodified turbine (ref. 3). Analogous tests were then conducted with coolant flow.

All turbine tests were conducted at a constant inlet stagnation pressure of 30 inches of mercury absolute (1.0159×10^5 N/m^2). Inlet temperature of both the primary and coolant air, as supplied by the laboratory combustion air system, was nominally 545° R (303 K).

SYMBOLS

A annular flow area, ft^2; m^2

g force-mass conversion constant, 32.174 ft/sec^2

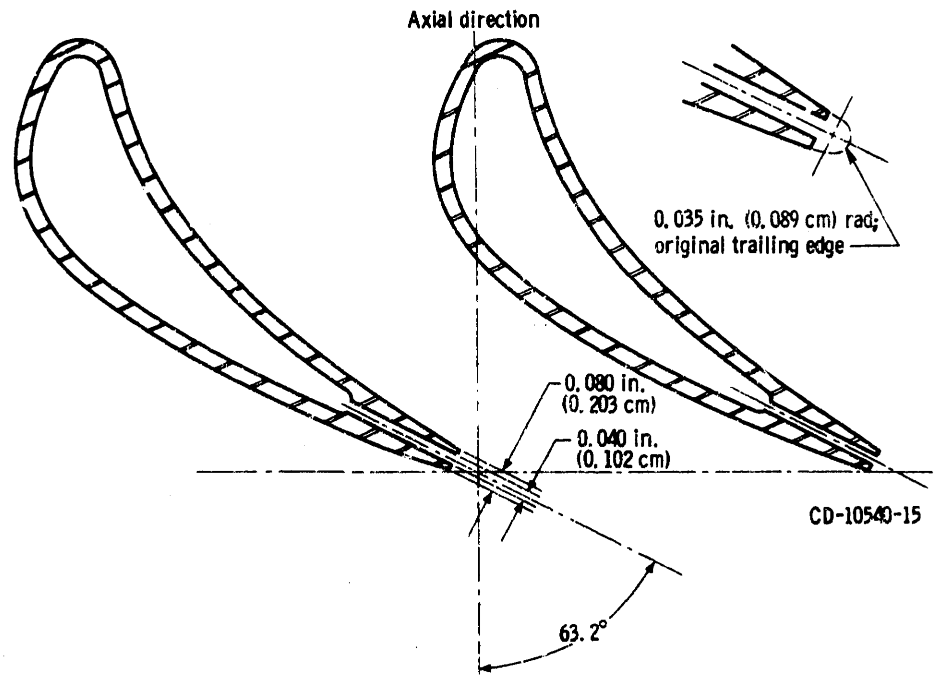
h	specific enthalpy, Btu/lbm; J/kg
J	mechanical equivalent of heat, 778.16 ft-lb/Btu
N	rotational speed, rpm
p	absolute pressure, lbf/ft^2; N/m^2
R	gas constant, 53.34 ft-lb/(lb)($^{\circ}$R); 287 J/(kg)(K)
T	temperature, $^{\circ}$R; K
w	mass-flow rate, lbm/sec; kg/sec
α	flow angle, measured from axial, positive in direction of rotor rotation, deg
γ	ratio of specific heats
δ	ratio of turbine inlet pressure to U. S. standard sea-level pressure
η	efficiency based on total-pressure ratio
θ_{cr}	squared ratio of critical velocity at turbine inlet to critical velocity of U. S. standard sea-level air
τ	torque, ft-lb; N-m

Subscripts:

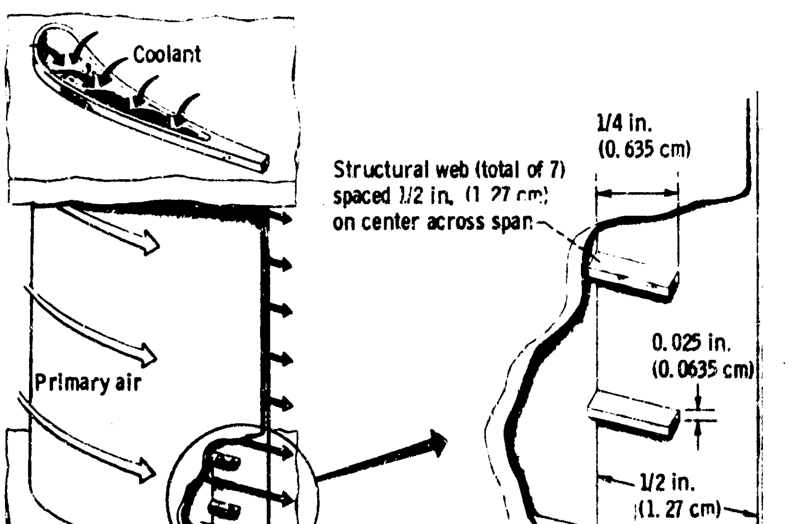
a	actual
c	coolant flow
cr	conditions at Mach 1
h	hub radius
i	measuring station at stator throat
id	ideal
p	primary flow
t	tip radius
th	thermodynamic
0	measuring station at turbine inlet (see fig. 6)
1	measuring station at stator outlet
2	measuring station at rotor outlet

Superscript:

'	total state



(a) Cross sectional view.



(b) Modified stator blade.

Figure 1. - Sketch of stator blade.

APPARATUS AND INSTRUMENTATION

A description of the turbine before modification, complete with stator and rotor blade coordinates, was presented in reference 4. This stator blading was modified as shown in figure 1. The rounded trailing edges were squared off and a coolant ejection slot cut through to the core. The slot extended 3.95 inches (10.03 cm) along a radial line from the stator hub to the stator tip and terminated 0.025 inch (0.0635 cm) from either shroud (fig. 1(b)). The slot had a width of 0.040 inch (0.102 cm) and an orientation angle of 63.2° (from the axial direction) both of which were constant along the radial line. The coolant entered the blade core from a coolant supply annulus located directly over the blades. A closeup photograph of the modified stator blade assembly installed in the facility is shown in figure 2.

The test facility described in references 3 and 4 was modified to include the stator coolant system. This system included the supply pipe from the combustion air header, an air-flow metering venturi, a control valve, the coolant inlet torus, the coolant supply annulus, and appropriate instrumentation for determining the coolant mass flow rate. Twelve $1\frac{1}{2}$-inch (3.81 cm) feeder pipes connected the torus to the supply annulus: These pipes, in conjunction with a circumferential baffle, assisted in distributing the air evenly over the outside circumference of the stator. A photograph of the modified stator assembly installed in the test facility with the outlet ducting removed is shown in figure 3.

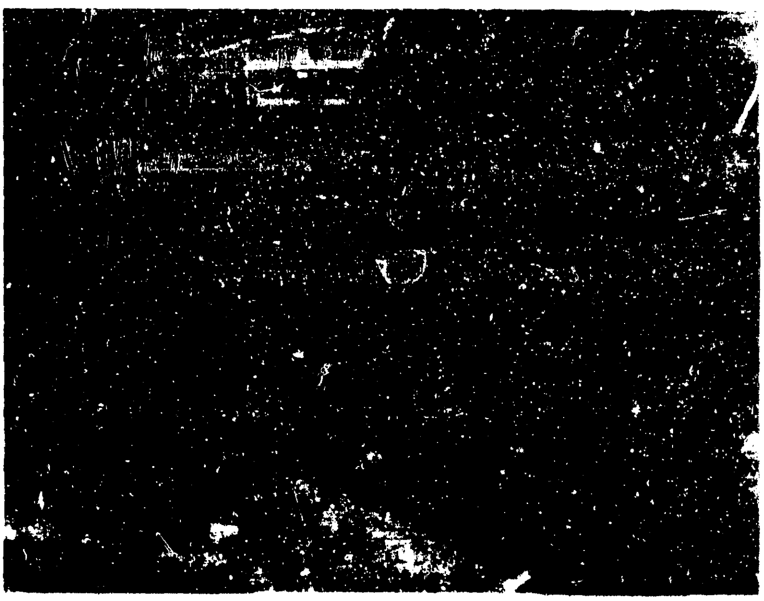


Figure 2. - Modified-cooled stator blades.

Figure 3. - Stator assembly installed in test facility.

A photograph of the test facility is shown in figure 4. The photograph was taken before the supply pipe to the inlet torus was installed. Both the coolant air and the primary air were supplied by the laboratory combustion air system and are at the same temperature, nominally 545° R (303 K).

The turbine rotor used in this performance evaluation was the same rotor used in the unmodified turbine described in reference 3. A photograph of the rotor is shown in figure 5.

The instrumentation was essentially the same as described in reference 3 except for that required to determine the coolant mass flow rate and its ideal energy. This instrumentation measured the venturi upstream pressure and differential pressure and the coolant-air temperature. Instrumentation was also provided to measure the temperature and pressure of the coolant at the coolant supply annulus.

The other instrumentation used was described in detail in reference 3 and measured the following: total and static pressures and temperatures at the turbine inlet, static pressures at the stator throat and stator exit, turbine exit static pressure, turbine exit flow angles, turbine speed, and torque. The measuring stations are shown in figure 6.

Figure 4. - Test facility.

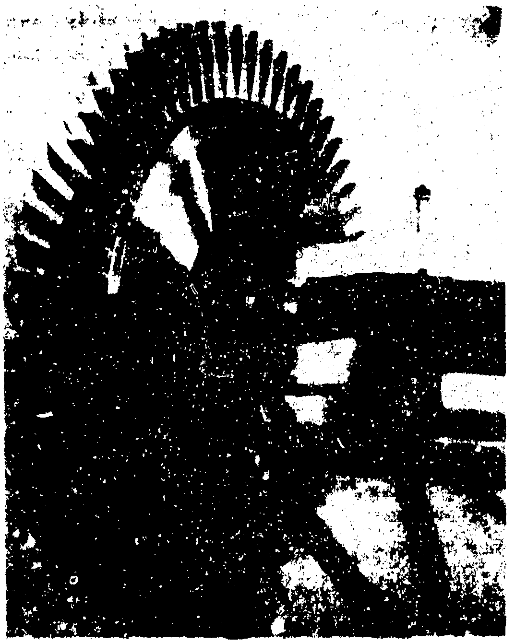


Figure 5. - Turbine rotor assembly.

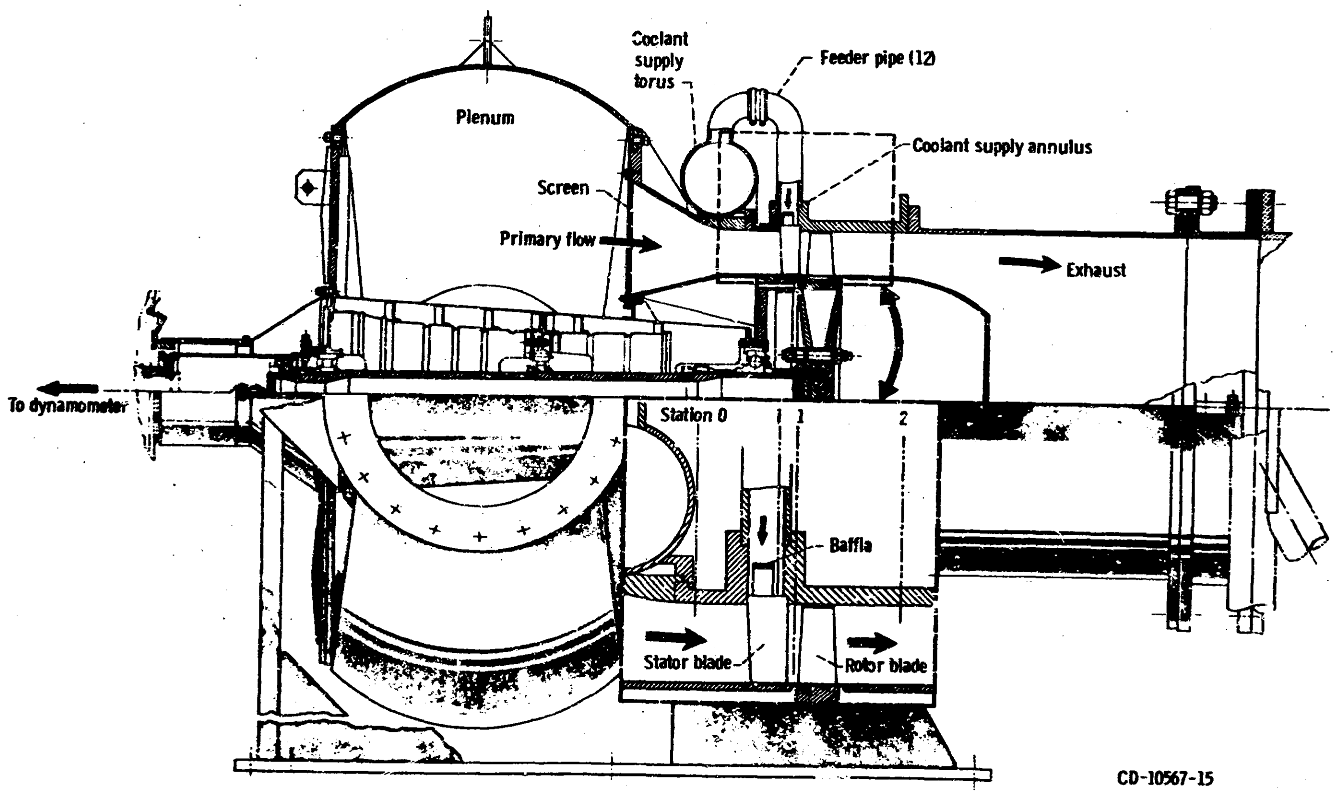


Figure 6. - Turbine test section.

All instrumentation was connected to a 100-channel data acquisition system which measured and recorded the electrical signals from the appropriate transducers. At every steady-state set point of turbine operation, five readings of each transducer were recorded. These readings were then averaged to determine the data.

PROCEDURE

The performance test for the modified turbine was conducted in three phases as follows:

(1) Performance data taken over a range of overall pressure ratio and speed with no cooling air ejection

(2) Performance data taken over a range of overall pressure ratio and speed with the cooling air supply annulus pressure equal to the turbine inlet pressure

(3) Performance data taken over a range of overall pressure ratio and coolant flows for design speed and approximately design work output

In all three phases the turbine-inlet total-state conditions were 30 inches of mercury absolute (1.0159×10^5 N/m^2) and approximately 545° R (303 K). The turbine overall pressure ratios were set by adjusting the turbine outlet pressure. At a given speed, the pressure ratio was varied from approximately 1.4 to the maximum obtainable. In phases (1) and (2) of the test procedure the speed was varied over a range of from 40 to 110 percent of design speed in 10-percent increments. For phase (2), the coolant-supply annulus pressure was maintained constant and equal to the turbine inlet pressure. In phase (3), coolant mass flow rate was varied by regulating the pressure of the coolant supply annulus. The regulation of this pressure resulted in coolant inlet pressures both below and above the turbine inlet pressure for the range of coolant flows reported herein and resulted in a range of coolant fractions (ratio of coolant to primary flow) of from zero to 0.07.

Turbine performance was based on total-pressure ratio. The inlet total pressure was calculated (as in ref. 3) from static pressure, mass flow, annulus area, and total temperature using the following equation:

$$\frac{p_0'}{p_0} = \left[\frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma}\left(\frac{w_p}{p_0 A_0}\right)^2 RT_0'}\right]^{\gamma/(\gamma-1)} \tag{1}$$

The outlet total pressure was calculated (as in ref. 3) using static pressure, turbine-exit flow angle, annulus area, and total temperature, but was modified to include the sum of the primary and coolant flows such that

$$\frac{p_2'}{p_2} = \left[\frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma}\left(\frac{w_p + w_c}{p_2 A_2}\right)^2 \frac{RT_2'}{\cos^2 \alpha_2}}\right]^{\gamma/(\gamma-1)} \tag{2}$$

The total temperature T_2' used in this equation was derived using inlet temperature, torque, mass flow, and speed data.

Two efficiencies are defined for use in this report: primary efficiency and thermodynamic efficiency. In equation form,

$$\eta_p = \frac{w_p \Delta h_p + w_c \Delta h_c}{w_p \Delta h_{id,p}} = \frac{\frac{\tau N \pi}{30J}}{w_p \Delta h_{id,p}} \tag{3}$$

$$\eta_{th} = \frac{w_p \Delta h_p + w_c \Delta h_c}{w_p \Delta h_{id,p} + w_c \Delta h_{id,c}} = \frac{\frac{\tau N \pi}{30J}}{w_p \Delta h_{id,p} + w_c \Delta h_{id,c}} \tag{4}$$

The primary efficiency, which relates the total power of both fluids to the ideal power of only the primary flow, is useful in engine cycle studies. The thermodynamic efficiency, which accounts for the ideal energies of both fluids, is useful in studying the loss characteristics of the fluids involved.

RESULTS AND DISCUSSION

The results of the experimental investigation are discussed in three phases. First, the overall turbine performance of the modified uncooled turbine is discussed and compared with that previously obtained for the unmodified turbine, that is, before the stator was modified for trailing-edge slot coolant ejection. Second, the turbine performance with cooling air supplied at a coolant annulus pressure equal to the turbine inlet pressure is discussed and compared with the modified-uncooled turbine performance. Third, the effect of varying amounts of coolant flow at the design speed will be discussed. Also included is a comparison of experimental results to those projected analytically.

Modified Uncooled Turbine Performance

The overall turbine performance is presented in terms of torque and mass-flow characteristics and the resultant performance map. Additional data include the flow angle at the turbine exit as a function of speed and overall total-pressure ratio. The static-pressure distribution at various measuring stations through the turbine is also presented for design speed and for a range of overall total-pressure ratio.

Overall turbine performance. - The overall performance map for the modified turbine with no coolant flow is presented in figure 7 in terms of equivalent specific work output $\Delta h/\theta_{cr}$ and a mass flow-speed parameter wN/δ for lines of constant total-pressure ratio p_0'/p_2' and equivalent rotor speed $N/\sqrt{\theta_{cr}}$. Contours of constant values of efficiency η, based on the total-pressure ratio across the turbine, are also included.

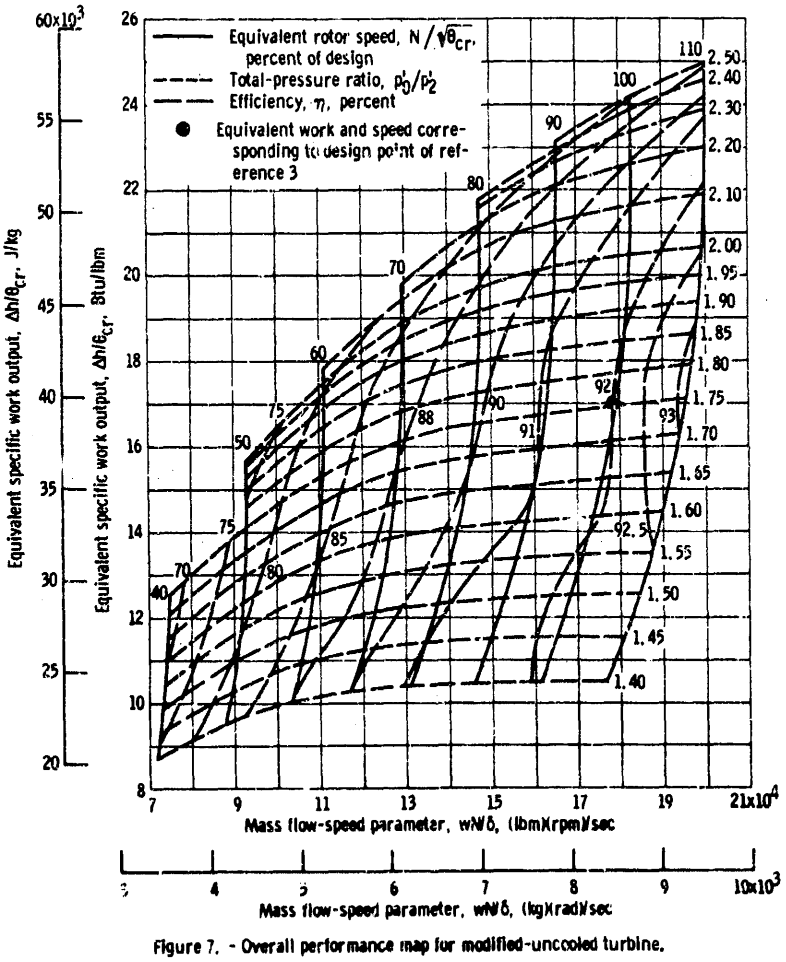


Figure 7. - Overall performance map for modified-uncooled turbine.

The turbine design specific work output of 17.00 Btu per pound (39 572 J/kg) at design speed is shown on the map (fig. 7). Design work was obtained at a pressure ratio of 1.755 and corresponds to an efficiency of about 0.920. This efficiency agreed very closely to the value of 0.923 obtained from the turbine (ref. 3) before the stator was modified. This agreement in efficiency is consistent with the stator tests of reference 2.

In general, the turbine yielded high efficiencies over a broad range of speed and pressure ratio. Maximum efficiencies of over 0.93 were obtained at 110 percent design speed and over a range of pressure ratio from 1.70 to about 1.86.

Torque and mass-flow characteristics. - The variation of equivalent torque τ/δ and of equivalent mass flow $w\sqrt{\theta_{cr}}/\delta$ with pressure ratio for the speeds investigated is shown in figures 8 and 9, respectively. Data from the faired curves of these two figures were used to calculate the performance map shown in figure 7.

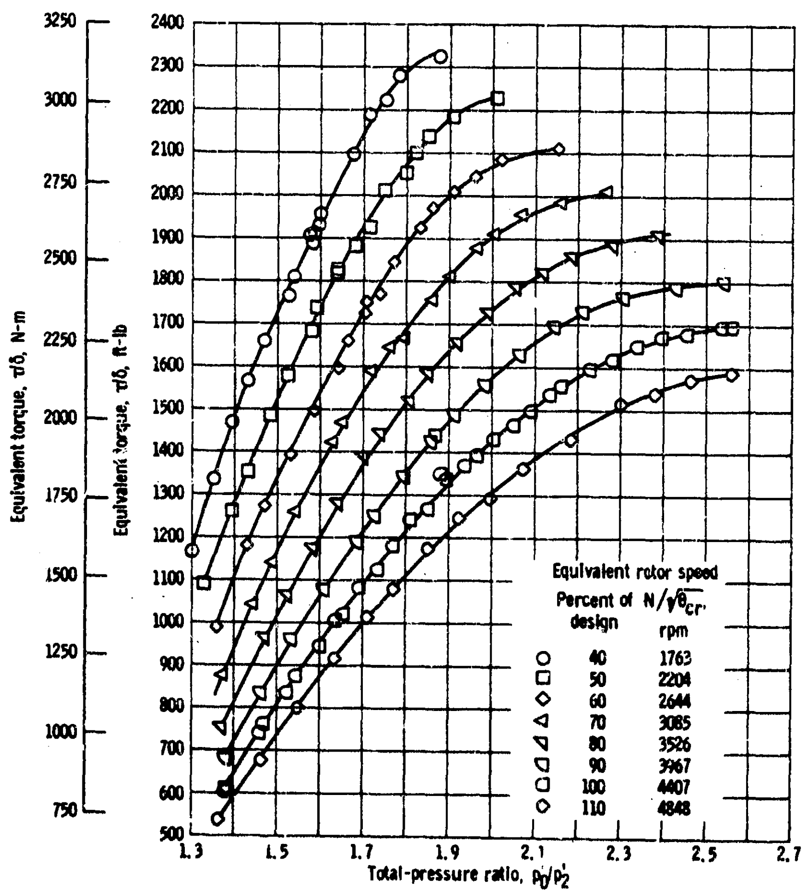


Figure 8. - Variation of equivalent torque with total-pressure ratio and equivalent rotor speed for modified uncooled turbine.

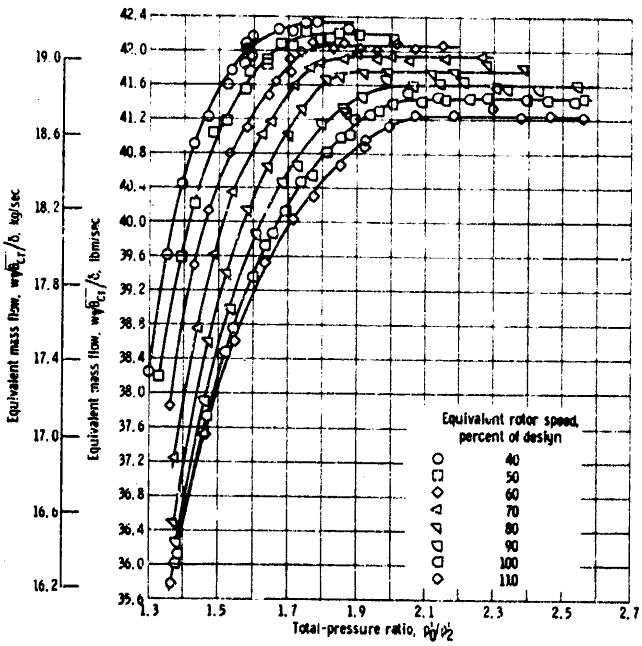


Figure 9. - Variation of equivalent mass flow with total-pressure ratio and equivalent rotor speed for modified uncooled turbine.

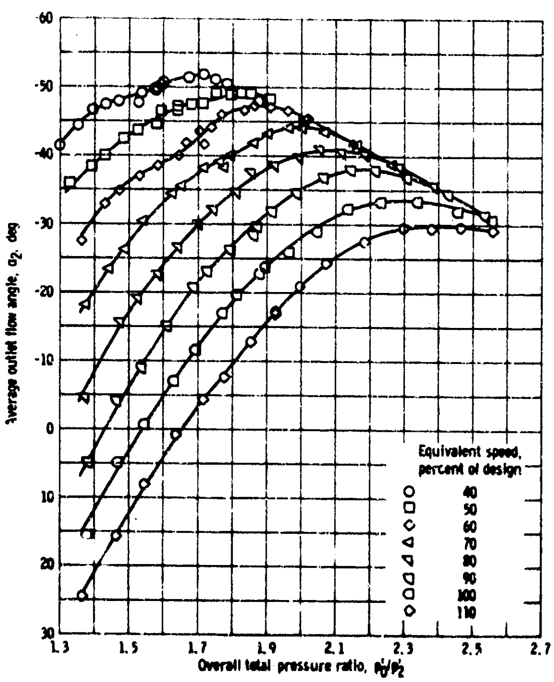


Figure 10. - Variation of outlet flow angle with total-pressure ratio and equivalent rotor speed for modified uncooled turbine.

The torque curves (fig. 8) are typical and show that torque continually increased with increasing pressure ratio for all speeds investigated. Limiting-blade loading, defined as that point where increases in pressure ratio result in no increase in torque output, was not attained at any speed.

The mass flow curves (fig. 9) show increasing flows with increasing pressure ratios for all speeds, until choked (constant) values are reached. The fact that the value of choked flow was different at different rotor speeds indicates that the rotor, and not the stator, limited the flow. At the point corresponding to design work output at design speed, the mass flow was 40.54 pounds per second (18.39 kg/sec). This agreed very closely to 40.64 pounds per second (18.43 kg/sec) as obtained from the unmodified turbine tests of reference 3.

Turbine-outlet flow angle. - The average flow angle at the turbine outlet is shown in figure 10 as a function of the speed and the total-pressure ratio across the turbine. Negative angle data corresponds to a positive contribution to turbine work output. The average angle indicated at conditions of design specific work output at design speed is -15.7°, occurring at a total pressure ratio of 1.755. This agreed closely to a value of -15.2° obtained from the unmodified turbine tests of reference 3. This agreement in angle is consistent with the close agreement in efficiency and flow between the two turbines.

Static pressure distribution. - The variation in static pressure through the modified uncooled turbine is shown in figure 11 as a function of total-pressure ratio for design speed. The static-pressure measurements at the hub are presented in figure 11(a); the tip measurements in figure 11(b). Choking in a blade row is indicated when the static pressure at the inlet to a blade row remains constant as the total-pressure ratio across the turbine is increased. Referral to figures 11(a) and (b) shows that the hub and tip sections of the rotor choked at approximately the same overall total-pressure ratio of 2.1. This is in agreement with the design-speed mass flow data presented in figure 9, which also shows that the rotor had choked at an overall pressure ratio of 2.1. The static pressure variations at other rotor speeds (not included) were similar.

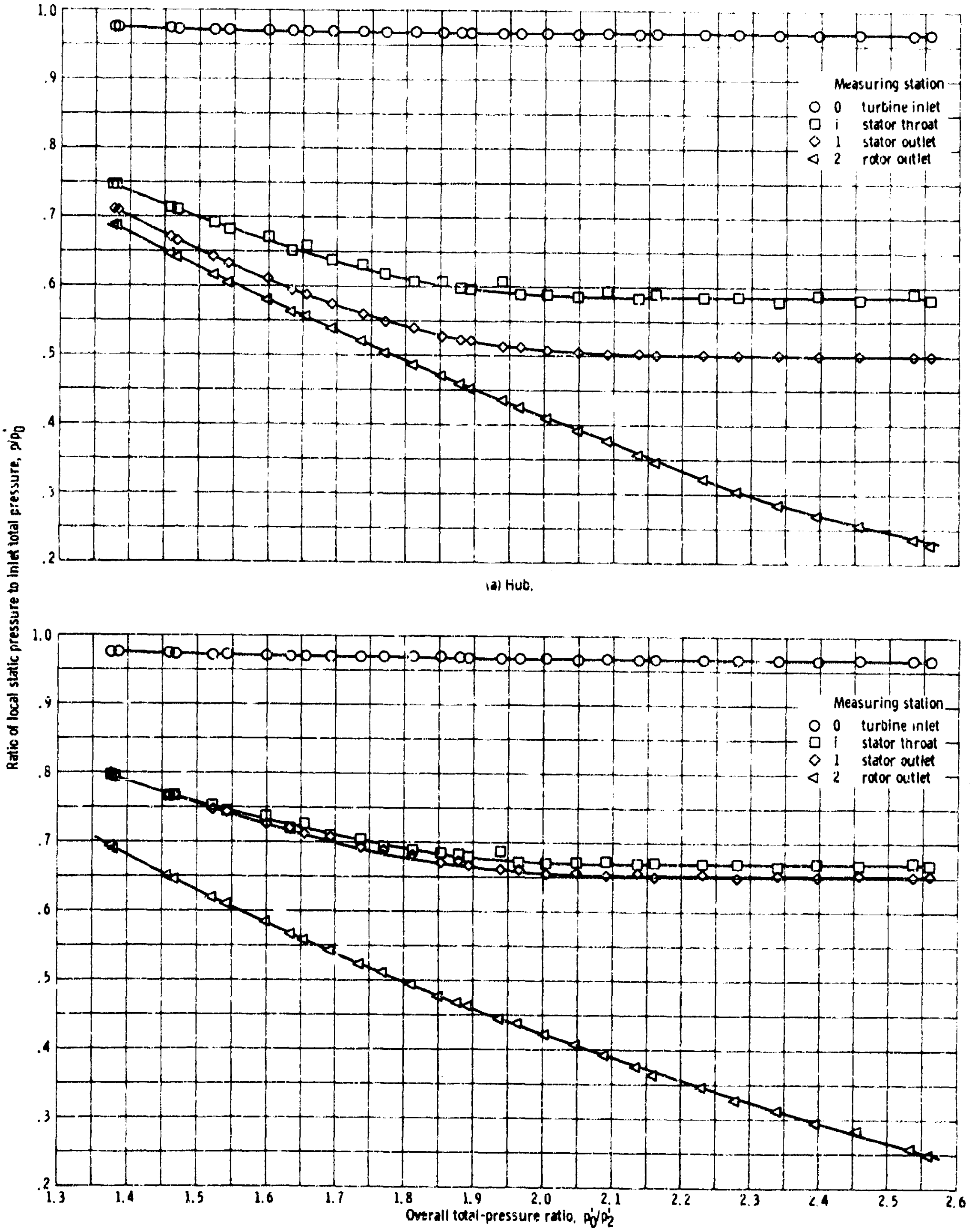


Figure 11. - Variation of static pressure through turbine with total-pressure ratio at equivalent design speed for modified uncooled turbine.

Modified Cooled Turbine Performance

These tests were conducted in the same manner as those previously discussed for the modified uncooled turbine, except for the addition of coolant air from the stator blade trailing-edge slots. For these tests, the total pressure in the coolant supply annulus over the stator blades (see fig. 6) was maintained constant at 30 inches of mercury absolute (1.0159×10^5 N/m^2), the same as the inlet total pressure to the turbine (primary air). This resulted in a coolant fraction w_c/w_p range from 0.0455 to 0.0482. These data are first presented in the same manner as for the uncooled turbine, and a comparison of the two configurations is made at the design work of 17.00 Btu per pound (39 572 J/kg).

Overall turbine performance. - The turbine performance map for this configuration is presented in figure 12. It is evident that the efficiencies are considerably higher than

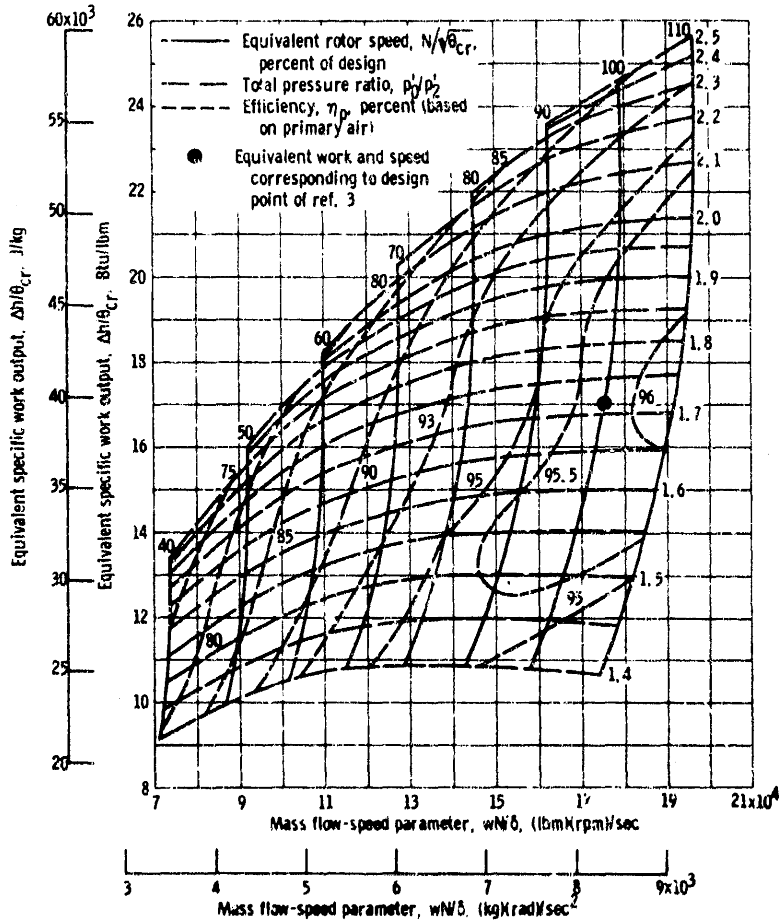


Figure 12. - Overall performance map for modified cooled turbine.

those obtained for the uncooled turbine (fig. 7). The efficiencies shown in figure 12 are the primary efficiencies as defined in the PROCEDURE section and calculated using equation (3). The increase in primary efficiency indicates an increase in torque caused by the addition of coolant flow. Values of efficiency greater than 0.96 can be noted at the 110-percent speed. At the design speed, at a specific work output of 17.00 Btu per pound (39 572 J/kg), the obtained efficiency was 0.958 occurring at a total-pressure ratio of 1.713. The attendant coolant fraction was 0.0468. The corresponding efficiency for the modified uncooled turbine was 0.920, occurring at a total-pressure ratio of 1.755.

Comparison of the cooled turbine performance map (fig. 12) with that of the modified uncooled turbine performance map (fig. 7) shows a similarity except for the efficiency levels. In the low-speed area of the maps, the cooled turbine shows about a five-point increase in efficiency. This efficiency difference decreased at the higher speeds. However, the areas of peak efficiency for both turbines occurred in the same general speed and pressure ratio regimes.

Torque and mass-flow characteristics. - The cooled-turbine performance map (fig. 12) was evolved from the torque and primary mass-flow data presented in figures 13 and 14, respectively. The turbine outlet flow angle data are presented in figure 15; the static-pressure distribution through the turbine for the design speed is shown in figure 16. These data have the same trends as found for the modified-uncooled turbine configuration (figs. 8 to 11) and are included for completeness.

Summary Comparison of Unmodified and Modified Turbines

Previous discussions have compared pertinent performance parameters of the modified uncooled turbine with those for the unmodified turbine. And the modified cooled turbine was compared with the modified uncooled turbine. These comparisons were made at a turbine operating point corresponding to design speed and a specific work output of 17.00 Btu per pound (39 572 J/kg) based on primary flow. These experimental data are summarized in the following table:

Turbine	Equivalent primary mass flow		Coolant fraction	Total pressure ratio	Outlet flow angle, deg	Primary air efficiency	Thermodynamic efficiency
	lbm/sec	kg/sec					
Design (ref. 3)	40.64	18.43	0	1.751	-15.2	0.923	0.923
Modified uncooled	40.54	18.39	0	1.755	-15.7	.920	.920
Modified cooled	39.72	18.02	.0468	1.713	-15.5	.958	.917

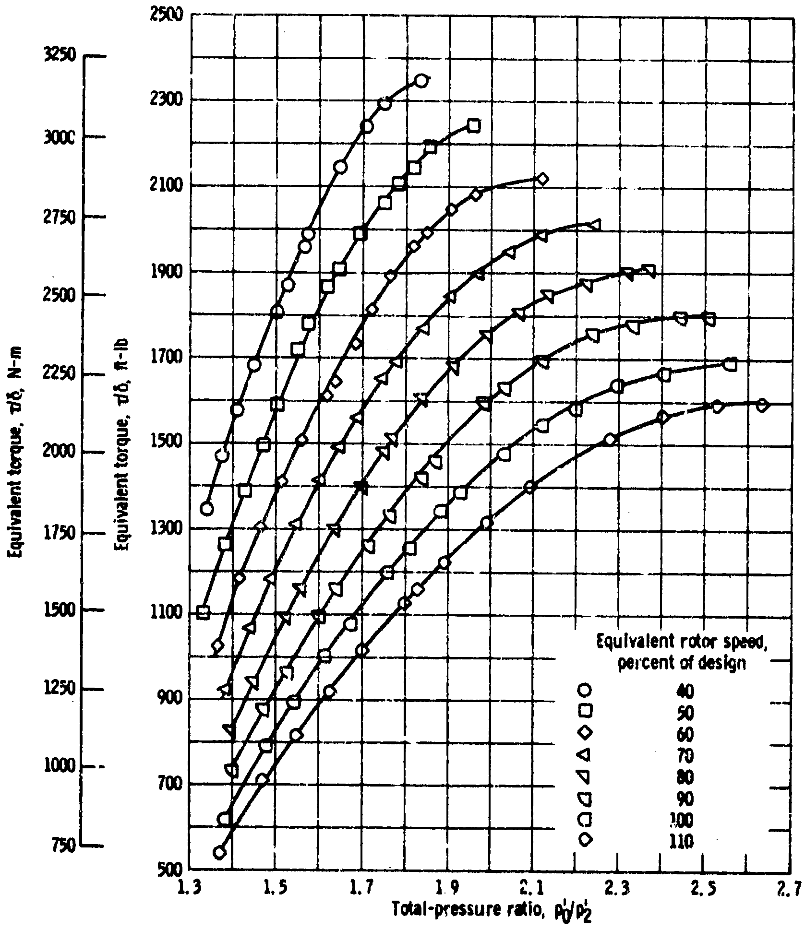


Figure 13. - Variation of equivalent torque with total-pressure ratio and equivalent rotor speed for modified cooled turbine.

These results show that no significant change in turbine performance occurred when the stator trailing edge was modified to include the coolant ejection slot. The addition of 0.0468 coolant fraction resulted in a 2-percent decrease in primary mass flow and corresponded to a net increase in total flow of about 3 percent. The rotor outlet flow angle was relatively unaffected by the coolant addition. Turbine efficiency, calculated on the basis of primary flow, increased about 3.8 points with the 0.0468 coolant fraction. This corresponds to an efficiency increase of 4.1 percent. The small change in thermodynamic efficiency (from 0.920 to 0.917), which accounts for the ideal energy of both the primary and coolant flows, indicates that trailing-edge coolant ejection had little effect on turbine performance.

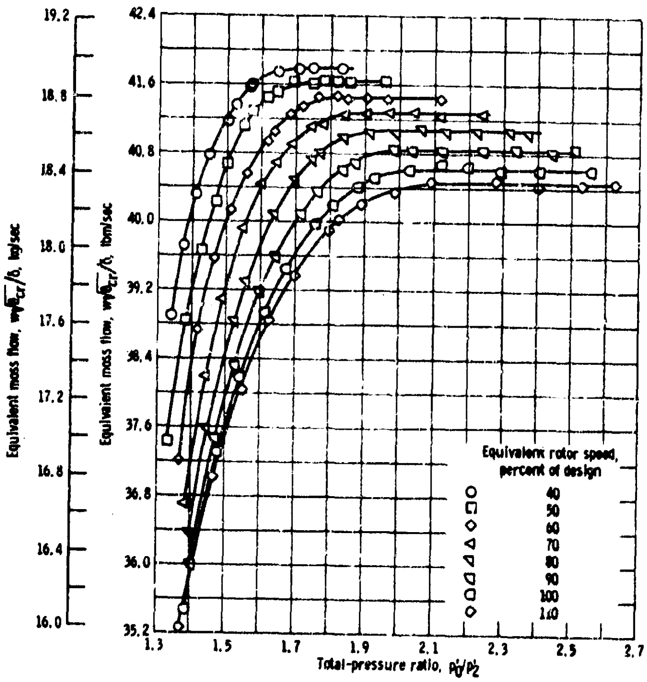


Figure 14. - Variation of equivalent mass flow with total-pressure ratio and equivalent rotor speed for modified-cooled turbine.

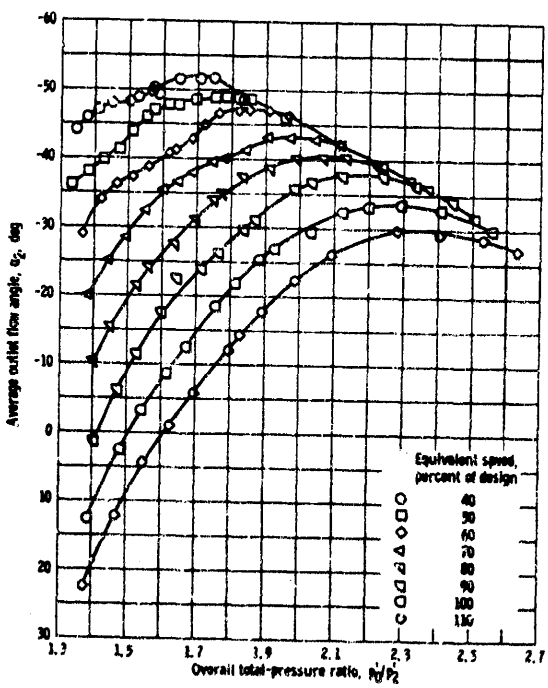


Figure 15. - Variation of outlet flow angle with total-pressure ratio and equivalent rotor speed for modified cooled turbine.

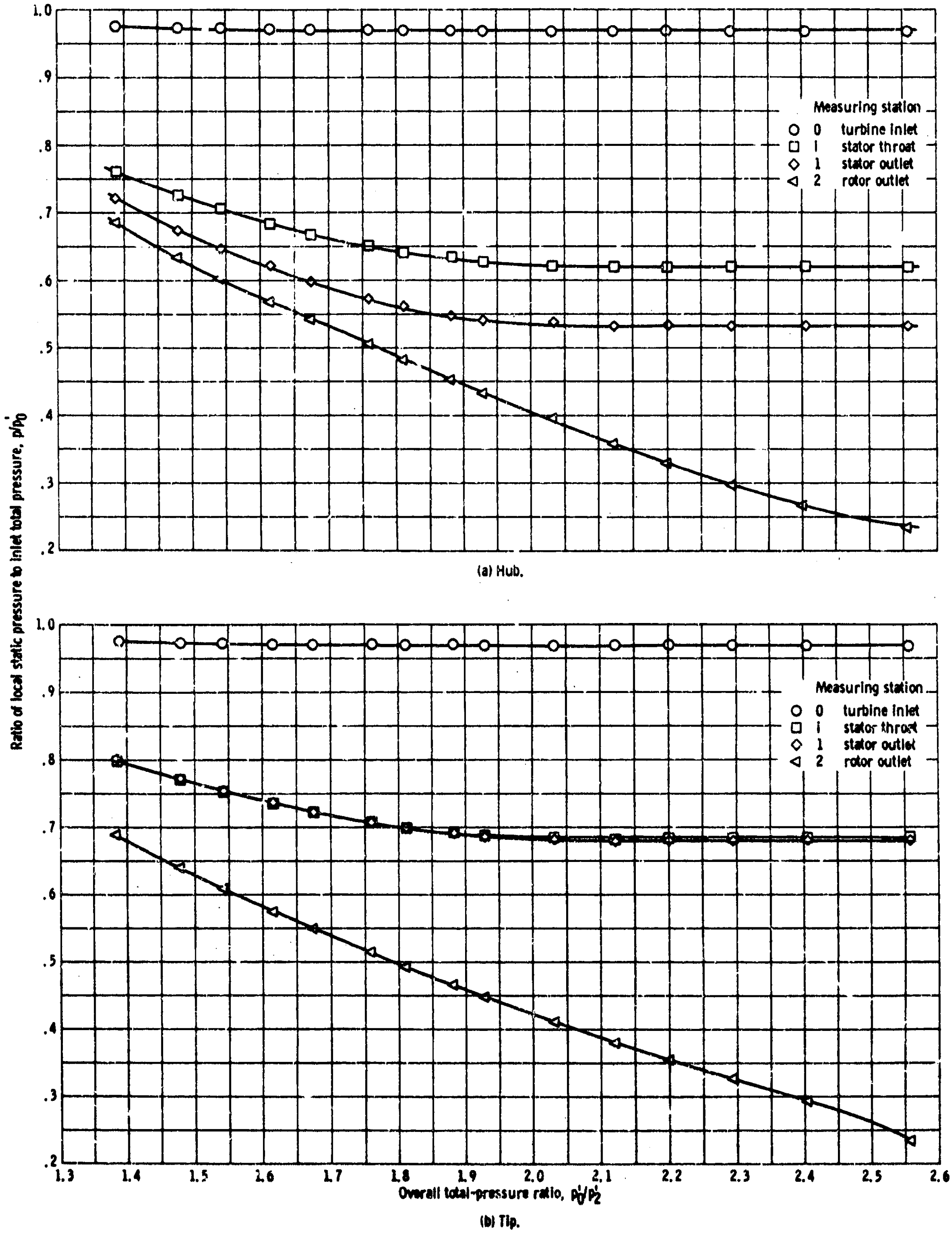


Figure 16. - Variation of static pressure through turbine with total-pressure ratio at equivalent design speed for modified-cooled turbine.

Effect of Coolant Flow on Turbine Performance

The effect of coolant flow on turbine performance was determined by investigating the turbine over a range of pressure ratios and coolant flow rates. The turbine was run at design speed and at pressure ratios bracketing the equivalent specific work output of 17.00 Btu per pound (39 572 J/kg). At each pressure ratio the coolant fraction was varied from zero to approximately 0.07.

From the experimental data, values of equivalent torque and mass flow as a function of total-pressure ratio for several values of coolant fraction were evolved. These parameters were then used to calculate the work output and efficiency as a function of coolant fraction.

Variation of flow and torque with coolant fraction. - Figure 17 presents the variation of equivalent primary flow, total flow, and torque as a function of coolant fraction at constant total-pressure ratio and design speed. The reference point at zero coolant fraction refers to conditions for the modified uncooled turbine and corresponds to a work output of 17.00 Btu per pound (39 572 J/kg). Figure 17 shows that with increasing coolant fractions, the total flow through the turbine rotor increased. The increase in total flow was less than the coolant flow because the primary flow concurrently decreased. For example, at the maximum coolant fraction of 0.07, the primary flow decreased

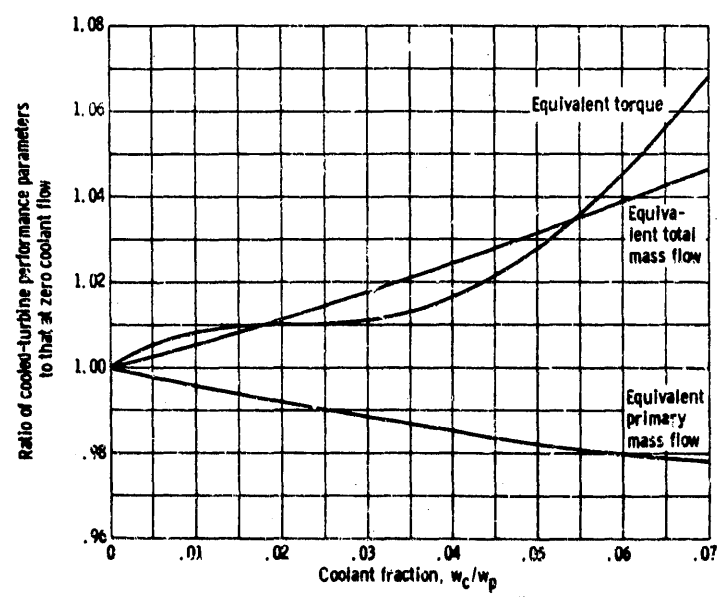


Figure 17. - Variation of equivalent torque, primary flow, and total flow with coolant fraction at constant overall total pressure ratio and equivalent design speed.

2.2 percent, and resulted in a net increase in total flow through the rotor of 4.6 percent.

The torque output increases with added coolant. The initial rapid rise in torque at very low coolant fractions results from a reduction in stator trailing-edge loss as the coolant fills the voids behind the blades. As more coolant was added, the energy of the coolant (inlet pressure) became higher relative to that of the primary air. This resulted, at coolant fractions above 0.054, in torque increases greater than total flow increases.

Variation of turbine efficiencies with coolant fraction. - Figure 18 shows the variation of primary and thermodynamic efficiencies as a function of coolant fraction at design speed and a work output of 17.00 Btu per pound (39 572 J/kg), based on the primary flow. Primary-air efficiency increased with coolant fraction with a trend similar to the torque variation of figure 17. At coolant fractions above 0.052, the gain in the primary efficiency was greater than the coolant fraction. For example, at 0.07 coolant fraction, the efficiency increased over 9 percent, which results from the inlet pressure (and specific energy) of the coolant being higher than the inlet pressure (and specific energy) of the primary air.

The thermodynamic efficiency first increases and then decreases with coolant fraction. The range of efficiency change is quite small (±0.007) which indicates that, for the turbine tested, there was little effect of coolant on thermodynamic performance. The reason for the trend of turbine thermodynamic efficiency can be directly related to the

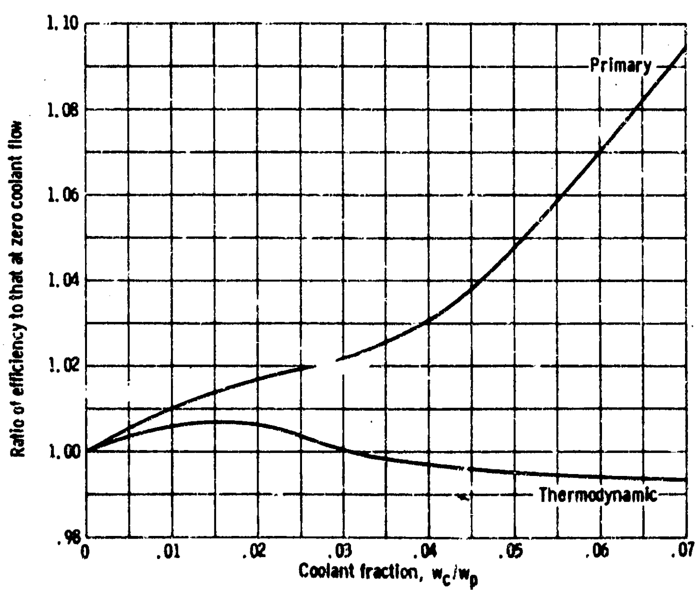


Figure 18. - Variation of efficiency with coolant fraction. Equivalent design speed; work output, 17.00 Btu per pound (39 572 J/kg); based on primary flow.

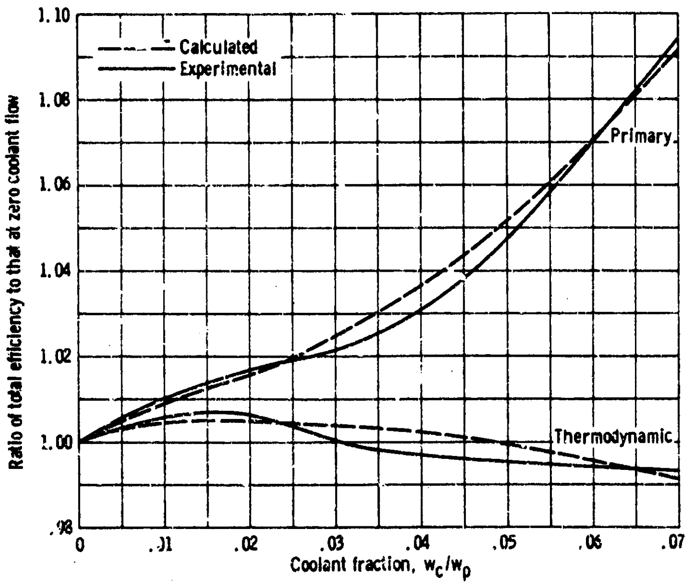


Figure 19. - Comparison of actual turbine efficiencies to those calculated from results of stator studies of reference 2. Equivalent design speed; work output, 17.00 Btu per pound (39 572 J/kg); based on primary air.

variation of stator efficiency with coolant fraction. This can be seen from figure 19, which superimposes the calculated variations in efficiency (dashed lines) based on the experimental results of reference 2, where the stator was tested separately. The calculated curves shown are based on the assumption that the only change in internal loss as a function of coolant occurs in the stator. It is concluded from the close agreement shown in figure 19 that the ejection of coolant through the stator trailing-edge slot had no significant effect on rotor performance. The addition of coolant had two counteracting effects. First, the frictional losses of the coolant within the blade increased with coolant flow and were always higher than the frictional losses of the primary air. This would cause a decrease in thermodynamic efficiency for all coolant fractions. Second, the presence of coolant from the slot into the wake decreased the trailing-edge loss of the primary air which would tend to increase the efficiency. At low flows (below 0.02), the recovery of trailing-edge loss predominates and results in efficiencies higher than that at zero coolant flow. As coolant fraction increases above 0.03, the frictional losses of the coolant predominates and results in thermodynamic efficiencies lower than that at zero coolant flow.

Comparison of experimental and predicted results. - The variation in primary work output with coolant fraction was predicted in reference 1 by two methods described in references 5 (isolated flow) and 6 (mixed flow). The primary work output $\tau N/w_p$ is

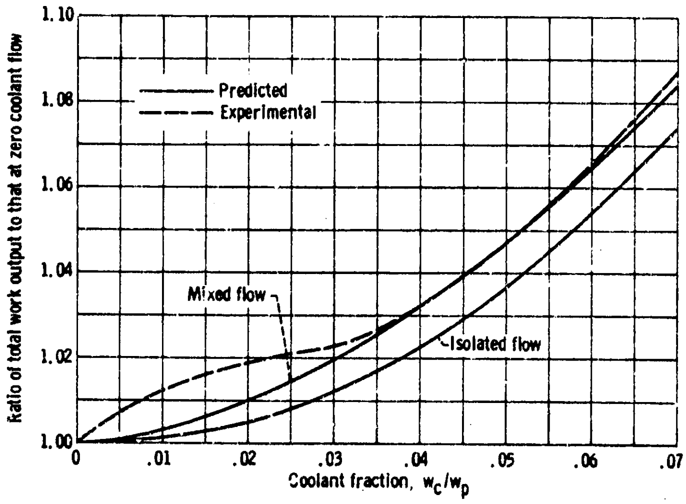


Figure 20. - Effect of coolant on primary work output at constant pressure ratio and design equivalent speed. Based on an equivalent specific work output of 17.00 Btu per pound (39 572 J/kg) at 0-percent coolant flow.

equal to the power output divided by the primary flow. The experimental variation was determined from the torque and primary-air mass-flow curves of figure 17. The comparison between the experimental and predicted variations is shown in figure 20.

The predicted performance variation for both the isolated flow and mixed flow procedures shown in figure 20 are similar, with both showing work output to increase with increasing coolant fraction. Agreement between the two predicted curves is within 1 percent. The variation in experimental work output agreed well with the results from the mixed-flow analysis. However, neither analytical method accounted for the reduction in stator blade trailing-edge wake loss.

SUMMARY OF RESULTS

A 30-inch (0.762-m) single-stage turbine was tested to determine the effect on performance of coolant ejection through slots in the stator trailing edges. The effect of the modification was determined by testing with zero coolant flow and comparing the results with those reported for the same turbine before the modification. The effect of coolant was studied by testing over a range of coolant fraction from zero to 0.07. The experimental effects of coolant addition were also compared with those predicted analytically. The major results are summarized as follows:

1. The stator blade modification had very little effect on turbine performance with no coolant flow. At design speed and work output, cutting off the round trailing edge re-

sulted in a turbine efficiency of 0.920. The turbine efficiency for the unmodified turbine was 0.923. This 0.920 value agreed with the referenced stator component tests, which indicated the stator efficiency to be virtually unaffected by the modification.

2. With a coolant supply pressure equal to the turbine inlet pressure, the modified cooled turbine performance map was similar to that for the modified uncooled turbine except for the efficiency levels. At design speed and a specific work output of 17.00 Btu per pound (39 572 J/kg), the modified cooled turbine efficiency (based on primary flow) was 0.958. The attendant coolant fraction was 0.0468.

3. The rotor performance was not significantly affected by the coolant over the 0.07 coolant fraction range tested. This was evidenced by the fractional change in overall turbine efficiencies being virtually the same as corresponding fractional changes in stator efficiencies over the complete range of coolant fractions investigated.

4. The primary-air efficiency, which relates the torque output to the ideal energy of only the primary flow, continuously increased with increasing amounts of coolant. Above coolant fractions of 0.052, the gain in primary-air efficiency was greater than that of the coolant fraction. At the maximum coolant fraction investigated (0.07), the efficiency increased more than 9 percent over that at zero coolant flow. This increasing rate of change reflected the manner of increasing coolant flow, that is, by increasing its inlet pressure (and specific energy) relative to that of the primary-air flow.

5. The thermodynamic efficiency, which relates the torque output to the ideal energies of both fluids involved, varied less than 1 percent (±0.007) over the 0.07 coolant fraction range tested. At low flows, the efficiency first increased due to a recovery of stator blade trailing-edge loss by ejecting the coolant into the wake. As the coolant fraction was increased, the increasing frictional losses of the coolant became predominant and resulted in a decrease in thermodynamic efficiency.

6. The experimental fractional change in work output with coolant fraction was approximated by two referenced prediction methods. The experimental fractional changes agreed well with those predicted.

Lewis Research Center,
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Cleveland, Ohio, December 3, 1969,
720-03.

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NASA-Langley, 1970 — 28 E-5418